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WADC TECHNICAL REPORT 53-391

A HERMETICALLY SEALED METAL-TO-METAL
MECHANICAL DRIVE

Y. C. SHEN

KEARFOTT COMPANY, INC.

SEPTEMBER 1953

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WRIGHT AIR DEVELOPMENT CENTER

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A HERMETICALLY SEALED METAL-TO-METAL
MECHANICAL DRIVE

Y. C. Shen
Kearfott Company, Inc.

September 1953

Electronic Components Laboratory
Contract No. AF33(038)-23311
RDO No. 111-82

Wright Air Development Center
Air Research and Development Command
United States Air Force
Wright-Patterson Air Force Base, Ohio

FOREWORD

The research and development work described in this report was performed under Contract AF33(038)-23311, Kearfott Project No. M-363, by Kearfott Company, Inc., Little Falls, New Jersey, and was undertaken for the purpose of improving the hermetically sealed mechanical drive originally developed under Contract W33-038-ac-21946. Messrs. H. D. Adkins and Y. C. Shen were the project engineers for the contractor.

The contract was initiated under RDO No. 111-82, PRESSURE TIGHT ROTARY DRIVE FOR PRESSURIZED ASSEMBLIES. The project was administered under the direction of the Electronic Components Laboratory, Directorate of Research, Wright Air Development Center, Wright-Patterson Air Force Base, Ohio. Messrs. I. S. Mayer, H. Silvestre, and L. V. McNamara were successively the project engineers for the Electronic Components Laboratory.

ABSTRACT

Development work on a metal-to-metal hermetically sealed mechanical drive, suitable for operation at 1800 rpm against a pressure differential of one atmosphere and capable of transmitting a torque of 25 inch-ounces was carried to completion as a continuation of an earlier project. The steps leading to the development of the final design of this drive are described in the present report. It is believed that the drive is suitable for quantity production and can be adapted to any specific requirements of a particular application within the limitations indicated in Exhibit TSELT-259, in accordance with which it was designed. More extensive testing under service conditions is recommended.

PUBLICATION REVIEW

The publication of this report does not constitute approval by the Air Force of the findings or the conclusions contained therein. It is published only for the exchange and stimulation of ideas.

FOR THE COMMANDER:

for *RE King*
RICHARD S. CARTER
Colonel, USAF
Chief, Electronic Components
Laboratory
Directorate of Research

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INTRODUCTION

Earlier development work performed in accordance with previous contract W33-038-ac-21496 was directed mainly toward the design and fabrication of a bellows suitable for a hermetically sealed mechanical drive. Under that contract the design of the drive was completed and a model was tested and found to be satisfactory. However, because of bellows failures, difficulties arose when attempts were made to duplicate results.

Work under the present contract was undertaken for the purpose of investigating the factors that affect the endurance strength of the bellows and of designing and building a number of producible models.

DEVELOPMENT OF BELLOWS ASSEMBLY TECHNIQUE

The original design of the bellows was based on the assumption that in order to obtain a bellows with the desired endurance properties, it was necessary to design for a maximum number of convolutions in a given space as well as for a maximum outside to inside diameter ratio. Since such bellows were not available, it was necessary to build them up from annular diaphragms. The diaphragms are formed from beryllium copper sheets, and are silver brazed to each other with the joints alternately on the outer and the inner periphery. Each joint is formed by folding the edge of one diaphragm around the edge of the mating diaphragm. The brazing is therefore done in two steps. First, the diaphragms are stacked in proper sequence; as they are stacked, a mixture of powdered silver brazing alloy and the brazing flux is applied to each joint; the stack is then introduced into a hydrogen atmosphere furnace and kept at 1450°F to accomplish the first brazing operation. The second brazing operation is effected in like manner after the edge of the diaphragms are rolled to form an intimate folded joint between each pair of diaphragms. The beryllium copper in the bellows thus assembled can be heat treated to the best strength by age hardening at 600°F for two hours.

In spite of the double brazing operation and the folded joints, it was found difficult to produce consistently sound joints at all the seams. Before the inception of this project, test models of bellows had been made in this manner. In fact, the bellows that lasted over 16 million cycles under contract no. W33-038-ac-21496, as well as the bellows that lasted over 17 million cycles were made in this way. (See paragraph 5, Life Test of Bellows under Tests Results).

The objective of this phase of the development was to devise such a design and to select such bellows materials that the performance of a successful bellows could be consistently reproduced in quantity and at low cost, when necessary. This led to experiments with clad materials.

Clad Material for Bellows

As described before, early models were made by stacking beryllium-copper diaphragms and applying a mixture of powdered brazing alloy and flux to the joints. This process was necessarily tedious. Besides, the rate of rejection was high because of insufficient silver at some parts of the joints.

Experiments were then conducted on a modified process, in which use was made of beryllium-copper sheets clad with a thin layer of brazing alloy metal. Although this procedure somewhat improved the situation and provided sufficient silver at the joints, life tests indicated that bellows made in this manner had rather low endurance (average of 200,000 cycles). A metallographic study showed that the cladding process, unless properly controlled, tended to introduce excessive boundary precipitate into the beryllium-copper structure. Consequently, the fatigue strength was considerably reduced.

Other difficulties were also encountered. Although the brazing alloy layer was kept to a minimum of 0.0005", the brazed joint tended to be excessively thick and seemed to be more vulnerable to repeated stress than the diaphragm material. As a result, the filler material in the joint always failed long before any damage occurred to the diaphragm material itself.

There was no evidence, however, that the silver alloy cladding in any way affected the strength of the beryllium copper.

Modified Assembly Technique

Experiments were performed with a simplified version of the earlier assembly technique. Instead of brushing a mixture of flux and silver alloy powder on the stack, the stack of beryllium-copper diaphragms was dipped in a thin paste made of the mixture. The dipped stack was then mounted in a fixture made of molybdenum and brazed in a furnace. The molybdenum fixture was sufficiently oxidized to keep the silver alloy from adhering to it during the furnace brazing operation. This method was found to give results equivalent to those of the brushing method. An absolute hermetic seal, however, could not be consistently obtained.

Other Bellows Material

Teflon Bellows: In line with the aim to survey all possible materials that might possibly serve the purpose, tests were performed on bellows machined from solid teflon material. The best record obtained on several models was 6,700,000 cycles. However, attempts to obtain a sure hermetic seal were unsuccessful. (The test models were merely clamped against rubber gaskets). As developed by the U.S. Gasket Co., Camden, New Jersey, the ends of the teflon material can be molded with a metallic (silver alloy) powder mixed with teflon powder. The mixture starts out with a small proportion of metal powder and ends with practically 100% metal on the outside surface. The soldered slug is then machined leaving two powdered metal ends for soldering. Tests showed, however, that the powdered metal ends cannot be made consistently hermetically sealed, even after tinning with solder. Furthermore, the metal ends tend to separate from the body of the bellows when the latter is under stress. The experiment was terminated without success.

Welded Bellows: Surveys were also made of other known methods of fabricating small bellows in order to check the feasibility of making a bellows of the desired dimensions by existing techniques. Manufacturers contacted were the Breeze Corporation, Newark, New Jersey, and the Bristol Co., Waterbury, Connecticut. All responses were in the negative.

Hydraulically Formed Beryllium-Copper Bellows: After extensive experiments with fabricated bellows, it became apparent that for high endurance, the ideal bellows would be a one-piece beryllium-copper bellows with a sufficient number of convolutions where the strengths of the joint and of the diaphragms are compatible. While earlier efforts to procure such bellows had been unsuccessful, consistent failure with silver soldered bellows renewed the effort. The Fulton Control Division of Robert-Shaw Fulton Co. was found to be in a position to supply a 14-convolution bellows of suitable inside and outside diameters.

Pending the arrival of this bellows, experiments were conducted with two 8-convolution beryllium-copper bellows made by Clifford Mfg. Co., joined in tandem, thus giving the effect of a 15-convolution bellows. Various types of joints were tried for the two bellows. Failure of the middle joint interrupted many life tests. The best record was 2 million cycles, obtained by using two bellows joined together by means of a brass ring-like adapter. No failures occurred in the bellows themselves.

Single, Hydraulically Formed Bellows Seal: The 14 convolution bellows was assembled in the sealed drive and given a life test. It withstood over 9 million cycles without failure. The design of the entire drive was thereupon accepted as satisfactory and work on the final engineering samples was started.

REDESIGN OF DRIVE COMPONENTS

In the early stages of this project, and concurrently with research work on bellows design, the other components of the sealed drive were reviewed and redesigned with a view to reducing the stress on the bellows to a minimum, without detriment to the bearings. This resulted in modifications in several parts. A set of new drawings are being submitted separately as remanufacturing data under Item C of the contract.

PREPARATION FOR TOOLING

Concurrently with the experiments on fabricated bellows, and in anticipation of the eventual possibility of quantity production, the design of suitable dies for the forming of the bellows diaphragms was studied. Two sources of die making were contacted, requirements discussed and tentative bids obtained on progressive dies. These dies were never procured however, since they were no longer needed.

LUBRICANT SELECTION

Under the previous contact, the Dow Corning Company had developed a special grease, designated as XG-101. However, upon more recent contact with this company for a new supply, we were informed that XG-101 was considered by them as unsatisfactory, and they recommended the DC-33 type lubricant.

Tests were also run on samples of grease submitted by Battenfeld Grease and Oil Corporation, N. Tonawanda, New York. Both samples, X-8414 and X-8353A, were found unsuitable for the purpose.

A number of samples of synthetic greases were tested. These samples are as follows:

- (1) "Ucon" Grease 818-Y (Exp. L-221) Lot No. 4523-56C
- (2) "Ucon" Grease 818-Y (Exp. L-30) Lot No. 4523-56B
- (3) "Ucon" Grease 818-Y (Exp. L-121) Lot No. 4523-34B
- (4) "Ucon" Grease 818-Y (Exp. L-60) Lot No. 4523-34A
- (5) "Bat's" Grease X8414
- (6) "Bat's" Grease X-8353-A

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Samples of greases 1 through 4 are made by Union Carbide and Carbon Corporation. Samples of greases 5 and 6 are made by Battenfeld Grease and Oil Corporation. Test results were as follows:

Samples 1 and 2: These greases contained an additive of molybdenum disulphide. These greases are not suitable for use in precision ball bearings, because particles of the additive prohibit smooth operation.

Samples 3 and 4: These samples contained the same greases as in 1 and 2, but graphite was used as an additive.

Samples 1, 2, 3 and 4 were subjected to a hot test by exposing them to a temperature of 175°C (347°F) in air. After a 2 hour period, the fluids in the greases had completely evaporated, leaving a residue of graphite or molybdenum disulphide.

Samples 5 and 6: These samples were subjected to the same test as in the preceding paragraph. After a 10 hour period, the materials reacted similarly and dry residues remained. In addition to a hot test, sample 6 was packed into a pair of standard R-10 ball bearings for a torque test. Results are tabulated below. For comparison, results previously obtained with the XG-101 silicone grease are likewise shown.

<u>Tests</u>	<u>Samples</u>	
	<u>K-8353-A*</u>	<u>XG-101**</u>
Breakaway Torque at 25°C (77°F)	.5 inch-ounce	.11-.13 inch-ounce
Breakaway Torque at -65°C (-85°F)	1.8 inch-ounces	.22-.26 inch-ounce
Running Torque at 1800 rpm and 25°C	5.0 inch-ounces	4.7 inch-ounces
Running Torque at 1800 rpm and 0°C	15.0 inch-ounces	6.1 inch-ounces
Running Torque at 1800 rpm and -65°C	25.0 inch-ounces	9.8-11.1 inch-ounces

TESTS PERFORMED AND TEST RESULTS

In order to evaluate the design against the requirements of Exhibit No. TSELT-259, the following tests were performed. Some of the tests were made on particular sub-assemblies while others were made on models of the drive.

(1) Leak Test.- A number of different tests were performed in order to test the seal. The bubble test under water was used as a quick check to locate leaks. The pressure was varied between 5 and 30 psi, depending

*As received.

**This sample had been exposed to a temperature of 200°C (392°F) for a period of 100 hours in air before test.

on the purpose of the test. The mass spectrometer leak detector was used whenever a final check of absolute hermetic sealing was made. The ability of a seal to hold a vacuum was checked during the life tests on the bellows.

(2) Speed.- A single set of bearings was used in all life tests and tests were run at various speeds up to 4000 rpm for an aggregate time of over several thousand hours. There were no signs of wear or malfunctioning in the bearings.

(3) Torque.- The same set of parts of a test unit of the Drive was repeatedly run under various loads up to rated load. A momentary torque of 200 inch-ounces was also applied statically for several periods of one second each. There were no signs of damage or malfunctioning of the component parts after the load was removed. All tests were made at room temperature.

(4) Salt Spray Test.- After a 50 hour salt spray test there were no signs of corrosion on the parts of a complete assembly of the drive.

(5) Life Test of Bellows.- In order to evaluate the endurance of the bellows seal against a pressure differential, the bellows was tested by flexing it in a drive housing sealed against a pressure of one atmosphere. The tests were conducted at room temperature, and the test equipment was so set up that a loss of seal would automatically stop the test and indicate its duration. Of the numerous bellows samples tested, the best record was made on a bellows assembly that withstood over 17 million cycles before a diaphragm ruptured, inspite of a very small imperfection in a brazed joint. The test model constructed in accordance with the final design was tested in this manner and after over nine million cycles, a leak developed in an inner edge of the bellows.

(6) Life Test of Bearings.- See under (2) and (3) above. When driven under full load, the small bearing originally used (U-10115) located at the oscillating end of the bellows developed failure in two units. The outer races of these bearings cracked in both cases. This bearing has since been changed to Miniature Precision Bearing No. 55-618, (Kearfott Dwg. No. 125441), which has a thicker outer race.

No test results are yet available at the time of this report. It is felt that the weakness of the previous bearing arrangement has been eliminated. Further testing had not yet been made at the time this report was written.

(7) Test on Bellows Material.- During the period of research for the purpose of better controlling the quality of the beryllium-copper material, various tests commonly known in the art were performed:

(a) The tensile strength of the raw materials was tested before and after various heat treatments designed to simulate the brazing operations, thus insuring that the basic strength of the material would not be lost during fabrication. Hardness readings could not be taken because of the extreme thinness of the material used.

(b) Metallographic inspection was also made of specimens of the raw materials as well as of the bellows diaphragms before and after processes to check the hardenability of the alloy.

(8) Cold Test.- The unit was soaked at an ambient temperature of -80°F for one (1) hour and checked for no-load friction torque. The torque was found to be approximately 4 inch-ounces as compared with 1.5 inch-ounces at room ambient at low speeds.

TESTS TO BE PERFORMED AND CHARACTERISTICS NOT TESTED

At the writing of this report the following tests were not performed, but it is believed that the drive of final design should be capable of satisfactorily meeting their requirements.

(1) Pressure Test.- The bellows used is capable of withstanding an internal or external pressure of 197 psi with its ends restrained according to the manufacturer's information. Therefore a pressure of 30 psi should be relatively insignificant in so far as the strength of the bellows is concerned. The soldered joints are so designed that deflection of the bellows has a minimum effect on the joint, which is mechanically spun over and does not rely on the filler material for strength.

(2) Storage.- The drive should be able to withstand storage in any position for periods of from 1 to 5 years without deteriorating, since the only cause of deterioration would be corrosion, and the unit passed the salt spray test satisfactorily.

(3) Vibration.- Existing production type sealed drives manufactured by Kearfott and the Type II Low Speed Sealed Drive developed for WADC and made with similar methods and materials have passed this type of test satisfactorily.

(4) Altitude.- This test is similar to the Pressure Test. A barometric pressure of 1.3 in. Hg. should not affect the performance of this drive.

(5) Humidity.-- See remarks under (3).

(6) Shock.-- See remarks under (3).

(7) Life Test of bellows and operation at hot and cold ambient temperatures.-- The physical strength of beryllium used in the bellows is not expected to change appreciably after extended exposure to high (200°C) and low (-65°C) temperatures. It is known that exposure of hardened beryllium copper to 200°C for 100 hours would reduce its ultimate strength by 2.5%, and chilling at temperature of -100°C actually increases its strength about 3.6% without embrittlement. The change in the bellows performance, therefore, should be negligible, if any.

Operation of the bearings at different temperatures is only limited by the properties of the bearing material and those of the lubricant. The 52100 steel used in the bearing has been given a high temperature draw to a hardness of Rc56. At 200°C (392°F) the hardness should not be affected. Neither would a low temperature of -65°C affect this material.

Exposure of the silicone lubricant DC-33 to temperatures of 200°C and -80°C did not apparently affect its consistency or the action of the bearing. The only change noticed was a slight discoloration (darkening) of the lubricant after subjection to prolonged heating.

TEST EQUIPMENT

A. Standard Equipment

- (1) Leak detector. General Electric Leak Detector, Catalog No. 6933950G3.
- (2) Salt Spray Equipment. Industrial Filter and Pump Mfg. Co. Type CAL.
- (3) Shock Tester. Berry Shock Mount Co. Type No. 150 VD Medium Impact Shock Tester.
- (4) Vibration Tester. For vertical direction: All American Tool & Mfg. Co. Model 10VA. For horizontal direction: Same manufacturer Model 10HA.

B. Special Equipment

(1) Bellows Life Test equipment. The endurance of the bellows sealing against a pressure differential of one atmosphere while being flexed in the drive, was tested with a bench arrangement, schematically shown in Fig. 1.

The mercury manometer served as a vacuum indicator as well as a switch, which turns off the test equipment and stops the clock when the vacuum is lost thus indicating a failure of the bellows. The motor speed was set

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constant at the desired value. The total number of cycles is shown by the lapse of time.

(2) Efficiency and torque test equipment.- The efficiency and performance with a torque load on the drive was tested on the bench setup schematically shown in Fig. 2. The housing of the drive was mounted on ball bearings so that friction torque could be measured directly by means of a dynamometer arrangement. Output load was applied by means of a prony brake. Spring scales were used for torque readings.

BEARING LOAD ESTIMATES

As reported under previous contract W33-038-ac-21496, the capacities of different bearings in the drive under normal and momentary loads, are as follows:

<u>Kearfott Part No.</u>	<u>Mfr's. No.</u>	<u>Normal Load</u>	<u>Momentary Load</u>	<u>Est'd Capacity</u>
X 111814 ***	SA955	70 lbs.	560 lbs.	**
X 111814	SA955	82.5 lbs.	660 lbs.	**
X 110115	S-154	7.8 lbs.	62.4 lbs.	50 lbs. *
X 110115	S-154	22.7 lbs.	181.6 lbs.	50 lbs. *
125441	SS-618	14.9 lbs.	119.2 lbs.	51.7 lbs. *
X 111816	SA956	12.5 lbs.	100 lbs.	**
X 111816	SA956	0 lbs.	0 lbs.	**

BELLOWS STRESS ESTIMATES

Stress due to pressure differential is expressed by the equations:

$$\text{At outer edge: } Sp = \frac{3w}{4t^2} \left[(a^2 - 3b^2) + \frac{4b^4}{(a^2 - b^2)} \left(\log_e \frac{a}{b} \right) \right] \quad (1)$$

$$\text{At inner edge: } Sp = \frac{3w}{4t^2} \left[(a^2 - b^2) - \frac{4a^2 b^2}{a^2 - b^2} \left(\log_e \frac{a}{b} \right) \right] \quad (2)$$

Where: Sp = Stress due to pressure differential (psi)
w = Pressure (psi)
t = Thickness of diaphragm (inch)
a = Major Radius (inch)
b = Minor Radius (inch)

In this instance:

w = 15 psi, under operation conditions.
w = 30 psi, for static pressure tests.
t = .004 inch. $t^2 = .000016$ inch
a = 0.4 inch. $a^2 = .16$ inch
b = 0.25 inch. $b^2 = .0625$ inch. $b^4 = .00391$ inch.
a/b = 1.6 $\log_e (a/b) = 0.47$

* Corrected for speed and life factors.

** No values can be given by manufacturer until more extensive test data become available.

*** Part numbers are Kearfott Numbers, and are arranged in order with reference to assembly drawing when viewed from left to right.

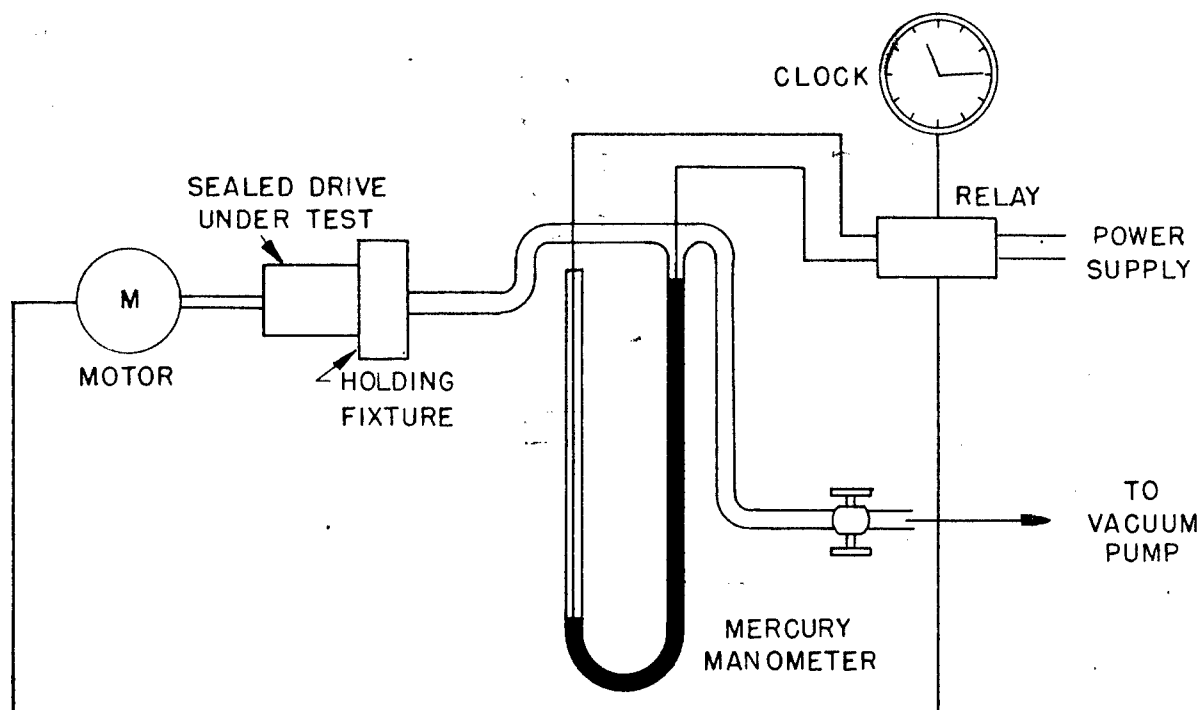


FIGURE 1

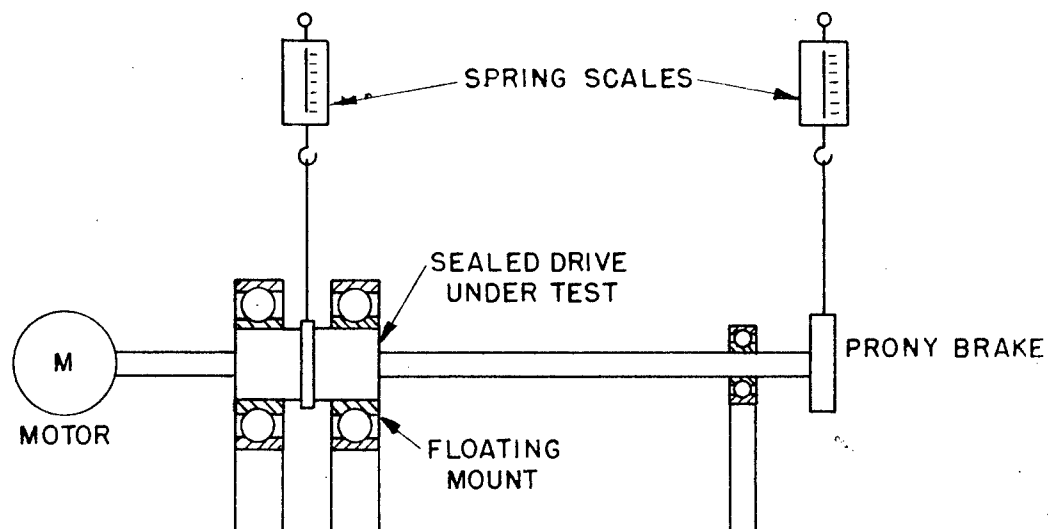


FIGURE 2

Therefore, at $r = a$, $S_p = 16,800$ psi and at $r = b$, $S_p = 20,700$ psi.

For static pressure tests, where $w = 30$ psi, the values of stresses due to pressure:

At outer edge: 39,600 psi.

At inner edge: 41,400 psi.

Flexing.- Stresses due to flexing of the bellows were determined by the following formulas:

$$\text{At } r = a: S_f = \frac{A_1 t E \theta}{a} \quad (3)$$

$$\text{At } r = b: S_f = \frac{A t E \theta}{b} \quad (4)$$

Where: S_f = Stress due to flexing (psi)

A_1 = Empirical constant.

A = Empirical constant.

t = Thickness, diaphragm, (inch)

E = Young's Modulus.

θ = Angle of flexure of each diaphragm in radians.

In this instance the constants for a ratio $\frac{b}{a} = 0.62$ are:

$$A_1 = 15$$

$$A = 22$$

$$E = 18 \times 10^6$$

$$\theta = \frac{15 \text{ degrees} \times 2\pi}{46 \times 360} = 0.0057 \text{ radians, each diaphragm.}$$

Therefore, at $r = a$, $S_f = 15,700$ psi and at $r = b$, $S_f = 22,500$ psi.

Axial Deflection.- Stresses due to axial compression or extension of the bellows in small magnitudes may be expressed as follows:

At $r = a$:

$$S_c = \frac{3w}{2\pi t^2} \left[1 - \frac{2b^2}{a^2 - b^2} \left(\log_e \frac{a}{b} \right) \right] \quad (5)$$

At $r = b$:

$$S_c = \frac{3w}{\pi 2t^2} \left[1 - \frac{2a^2}{a^2 - b^2} \left(\log_e \frac{a}{b} \right) \right] \quad (6)$$

Where: S_c = Stress due to compression or extension (psi)

w = Deflecting force (pounds)

t = Thickness of diaphragm (inches)

At $r = a$: $S_c = 12,000$ W and at $r = b$, $S_c = 16,000$ W.

It is evident that, at one time, the stresses are compressive on one edge and tensile on the other edge.

Summation of Stresses.- Calculations show that, with a slight axial compression in the bellows at assembly, it is possible to obtain working stresses that permit more evenly distributed factors of safety at the edges. Taking any point on the outer edge of one surface of a diaphragm, and a corresponding point on the inner edge along the radius on the same surface of the diaphragm, stresses at these points can be tabulated and summed up as follows: (tensile stress shown by a "plus" sign, and compressive stress shown by a "minus" sign.)

<u>Crank Position</u>	<u>Stresses</u>	
<u>Degrees</u>	<u>At Outer Edge</u>	<u>At Inner Edge</u>
0	$S_c = 0$ psi $S_p = -16,800$ psi $S_f = \underline{+15,700}$ psi $-1,100$ psi	$S_c = 0$ psi $S_p = -20,700$ psi $S_f = \underline{+22,500}$ psi $+1,800$ psi
90	$S_c = 0$ psi $S_p = -16,800$ psi $S_f = \underline{0}$ psi $-16,800$ psi	$S_c = 0$ psi $S_p = -20,700$ psi $S_f = \underline{0}$ psi $-20,700$ psi

The stresses vary, and make two complete cycles for each revolution of the crank, between $-1,100$ psi and $-16,800$ psi on the outer edge and between $+1,800$ psi and $-20,700$ psi on the inner edge. It should be noted that the varying stresses are not completely reversed. To estimate the factors of safety, an empirical formula may be used:

$$S_{max.} = S_e \frac{(3)}{2 - r}$$

Where: $S_{max.}$ denotes the endurance limit for the particular stress cycle.

S_e represents the endurance limit for complete reversed stress.

r is the ratio of minimum stress to maximum stress during a cycle, with r negative for partially or completely reversed stress

In this instance:

$$r \text{ is } \frac{-1700}{-16800} = +.0066, \text{ and } \frac{+1800}{+20700} = -.0847, \text{ at}$$

outer and inner edges, respectively.

The endurance limit for beryllium copper under completely reversed stress is taken to be 35,000 psi. Therefore, S_{max} and the factors of safety in the case of a bellows flexing under a pressure differential, but free from axial extension or compression, are obtained as follows:

<u>At outer edge</u>	<u>At inner edge</u>
$S_{max}: 35,000 \times \frac{3}{2-.006} = 52,400 \text{ psi}$	$35,000 \times \frac{3}{2+.087} = 50,400 \text{ psi}$
Factor of Safety: $\frac{52,400}{16,800} = 3.12$	$\frac{50,400}{20,700} = 2.43$

For the case of a bellows with a 0.25 pound axial compression, the same estimate is made, with results as tabulated below. It appears that a compressive preload tends to alleviate the stress on the inner edge, at the cost of slightly increasing the stress at the outer edge:

<u>Crank Position</u>		<u>Stresses</u>	
<u>Degrees</u>	<u>At Outer Edge</u>	<u>At Inner Edge</u>	
0	$S_c = -3,000 \text{ psi}$	$S_c = +4,000 \text{ psi}$	
	$S_p = -16,800 \text{ psi}$	$S_p = -20,700 \text{ psi}$	
	$S_f = +15,700 \text{ psi}$	$S_f = +22,500 \text{ psi}$	
	$-4,100 \text{ psi}$	$+5,700 \text{ psi}$	
90	$S_c = -3,000 \text{ psi}$	$S_c = +4,000 \text{ psi}$	
	$S_p = -16,800 \text{ psi}$	$S_p = -20,700 \text{ psi}$	
	$S_f = 0 \text{ psi}$	$S_f = 0 \text{ psi}$	
	$-19,800 \text{ psi}$	$-16,700 \text{ psi}$	
r:	$-4,100 = .207$	$+5,700 \text{ psi} = .342$	
	$-19,800$	$-16,700$	
$S_{max}: 35,000 \times \frac{3}{2-.207} = 58,400 \text{ psi.}$		$35,000 \times \frac{3}{2+.342} = 44,800 \text{ psi}$	
Factor of Safety: $\frac{58,400}{19,800} = 2.95$		$\frac{44,800}{16,700} = 2.68$	

CONCLUSIONS

(1) From limited test results available at this writing the development work performed under the present contract led to a basic design which should substantially fulfil the requirements of Exhibit No. TSELT-259. Specifically, the following

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tests were successfully passed:

- a. Leak test
- b. Speed
- c. Momentary torque
- d. Bellows life
- e. Salt spray

The following tests were not made at this writing, but the design should be capable of meeting the requirements:

- a. Pressure test
- b. Storage life
- c. Vibration
- d. Altitude
- e. Shock
- f. Efficiency

The following characteristics require more extensive tests in order to be evaluated:

- a. High and low temperature operation
- b. Bearing life

(2) From the basic design, further modifications may be made when more extensive tests are made and as more specific design requirements for particular applications are determined.

(3) More exhaustive tests should be made on the drive to better establish its limitations under various combinations of service conditions.

(4) Although all possible applications of this drive cannot be predicted, it may be desirable to standardize on a number of sizes and types of mechanical sealed drives which can be adapted to various applications. By drawing from "stock" drives, it would not then be necessary to design special drives for each installation.

(5) Additional tests performed by the Electronic Components Laboratory on submitted models failed to substantiate tests performed by the contractor. Inspection of disassembled models uncovered details that make results of many tests performed inconclusive.

(Inserted by the Electronic Components Laboratory.)

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